United States Patent [19]

Menke et al.

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[54] **JOURNAL BEARING**

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[52] **U.S. Cl.** **308/134.1;** 308/9; 308/121

[51] **Int. Cl.²** **F16C 1/24**

[58] **Field of Search** 308/121, 127, 133, 134.1, 308/9

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EXEMPLARY CLAIM

1. An improved journal bearing comprising in combination a non-rotatable cylindrical bearing member having a first bearing surface, a rotatable cylindrical bearing member having a confronting second bearing surface having a plurality of bearing elements, a source of lubricant adjacent said bearing elements for supplying lubricant thereto, each bearing element consisting of a pair of elongated relatively shallowly depressed surfaces lying in a cylindrical surface co-axial with the non-depressed surface and diverging from one another in the direction of rotation and obliquely arranged with respect to the axis of rotation of said rotatable member to cause a flow of lubricant longitudinally along said depressed surfaces from their distal ends toward their proximal ends as said bearing members are rotated relative to one another, each depressed surface subtending a radial angle of less than 360°, and means for rotating said rotatable bearing member to cause the lubricant to flow across and along said depressed surfaces, the flow of lubricant being impeded by the non-depressed portions of said second bearing surface to cause an increase in the lubricant pressure.

5 Claims, 5 Drawing Figures

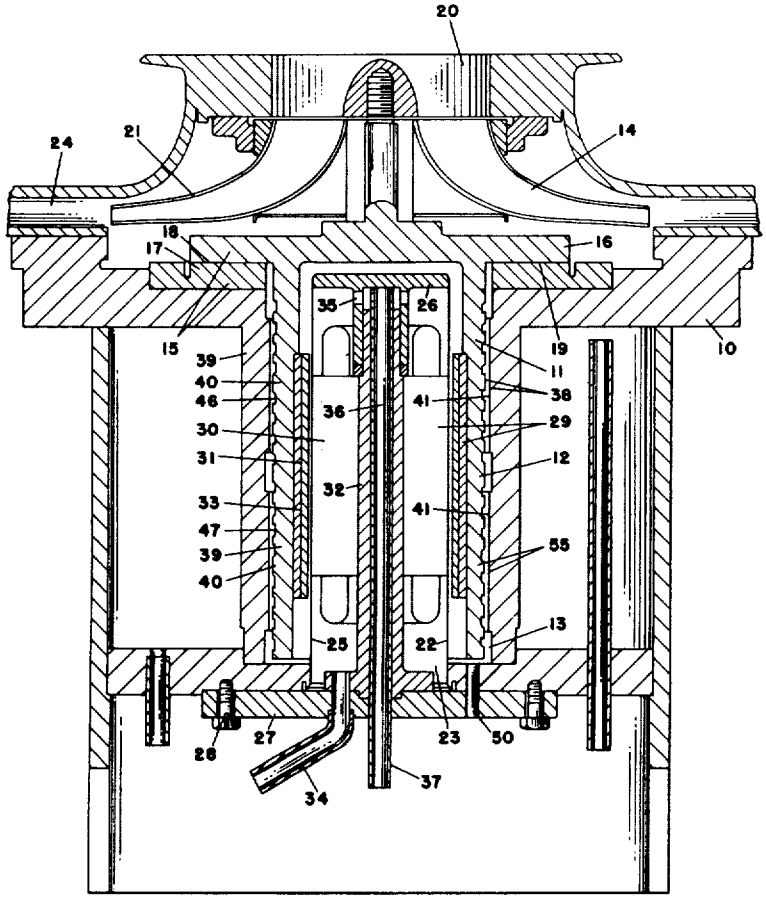

Fig. 1.

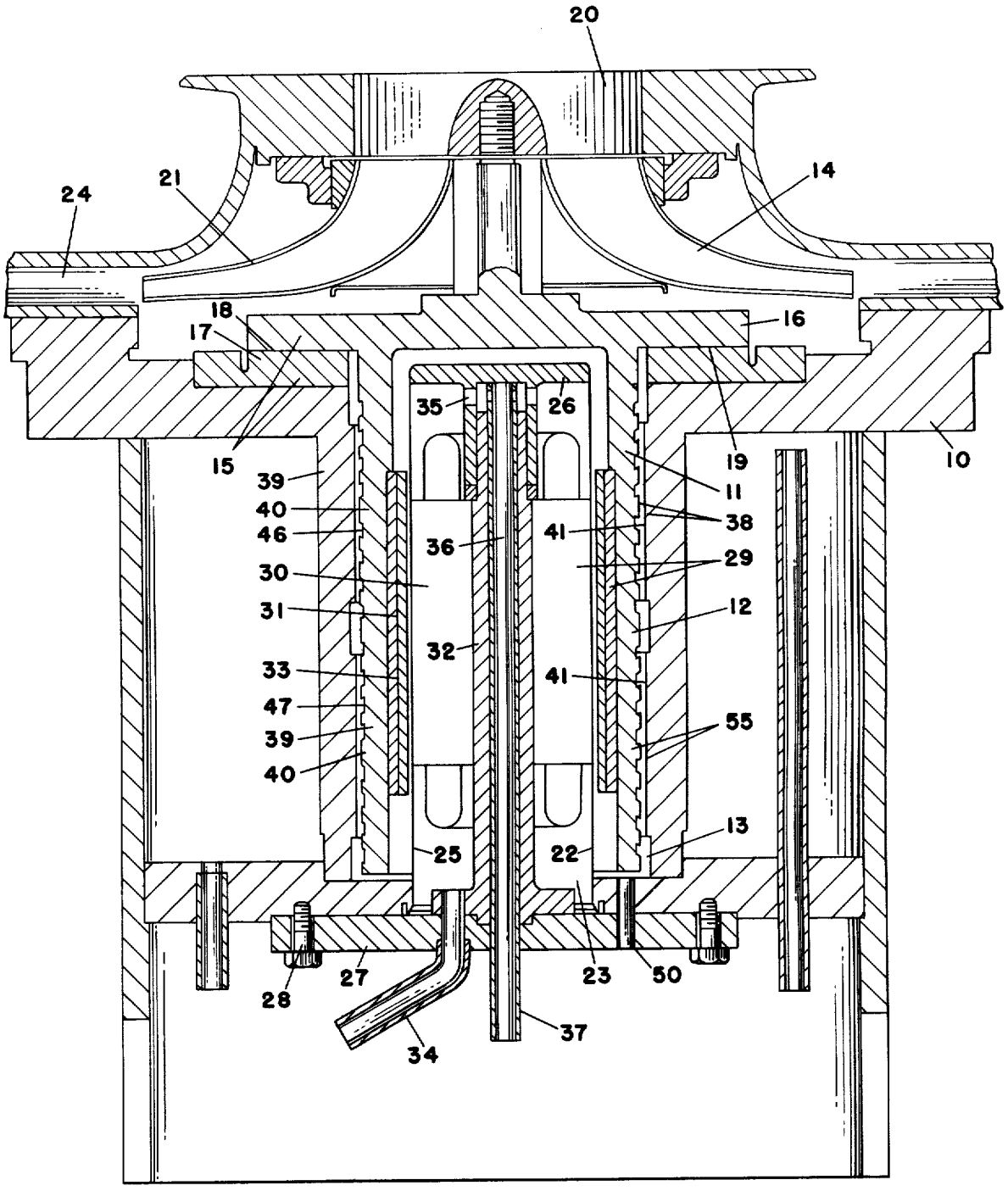

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Fig.2.

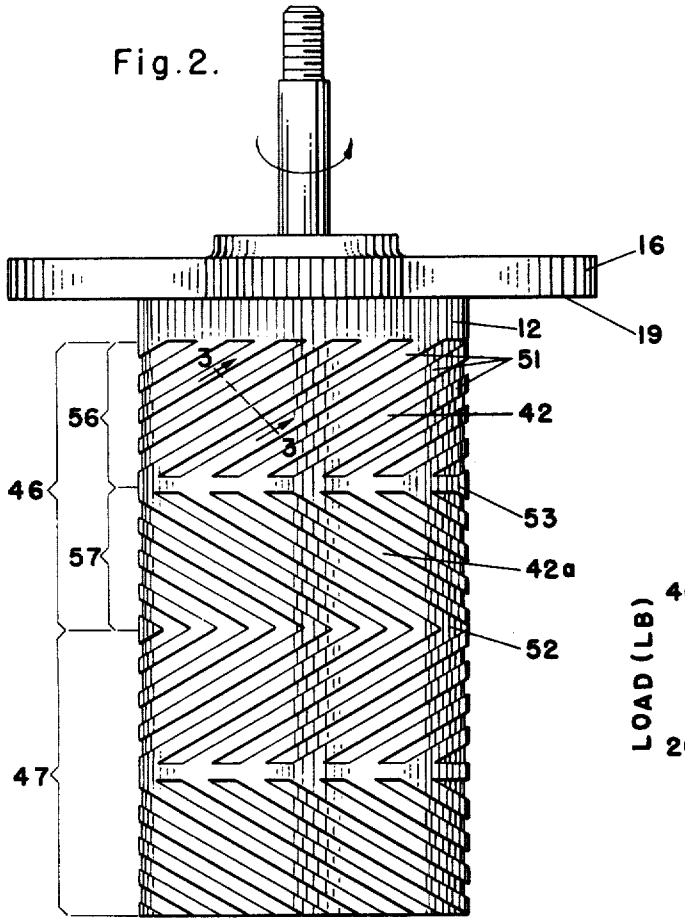


Fig.3.

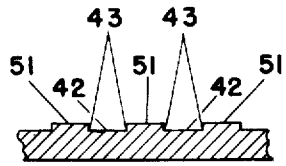


Fig.4.

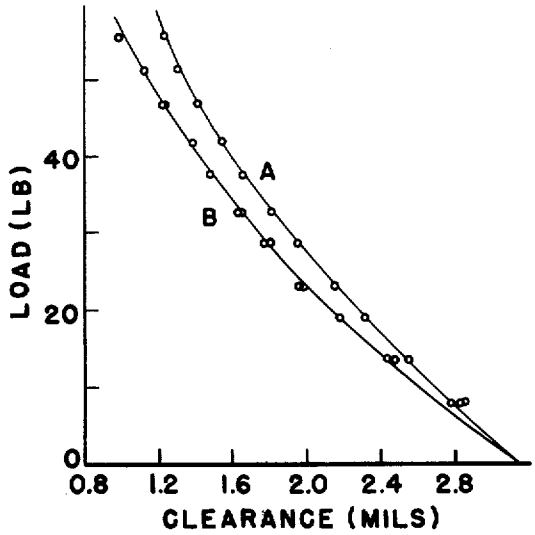
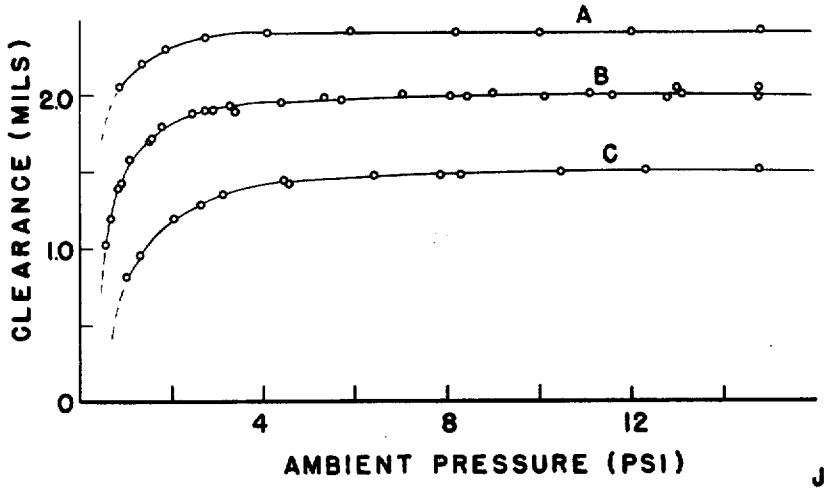


Fig.5.



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JOURNAL BEARING

This invention relates to a journal bearing particularly suited to the use of gaseous lubricants and more particularly the invention is concerned with a journal bearing whose confronting surfaces incorporate a novel structure designed to improve the load-carrying capacity and stability of the bearings.

Conventionally lubricated bearings are subject to a number of disadvantages and limitations. For example, in cases where a bearing is contained in a pump or compressor handling a corrosive gas, the corrosive gas often reacts chemically with the lubricant and destroys its lubricating properties. A further disadvantage of conventional lubricants is that the gas handled by the pump may become contaminated either with the lubricant or with the products of the reaction between the corrrosive gas and the lubricant. On the other hand, if the corrosive gas being handled by the pump is itself to be used as the lubricant, both of these difficulties can be readily overcome.

Another important factor in choosing a lubricant is the effect of the lubricant in reducing the friction between the confronting bearing surfaces. An important advantage of gaseous lubricants is that the frictional effects are considerably reduced when compared to the frictional effects of liquid or solid lubricants. This, of course, is the result of the relatively low viscosity of gases. The power saving derived from the use of gaseous lubricants may often be as high as 99% for the same bearing area.

These clear advantages, though of primary consideration, have not led to a wide utilization of gas-lubricated bearings. Until just a short time ago, the industrial usage of gas-lubricated bearings was quite rare, since it was thought that such bearings could not support a sufficiently large load to be used successfully.

Recently, however, a number of successful thrust bearings suited for gaseous lubricants have been developed. Most of these gas-lubricated thrust bearings are based on the action of especially fabricated bearing surfaces which act to pump the lubricant from the outer periphery of the bearing inward toward the rotating shaft. The pressure of the lubricant is increased sufficiently by this pumping action to support and lubricate a rotor having a considerable mass. A typical thrust bearing of this type is disclosed in the co-pending application of John R. Dunning et al., Ser. No. 683,872, filed July 16, 1946. The thrust bearing disclosed in that application comprises a first bearing member having a plane bearing surface and a second bearing member having a confronting bearing surface provided with a plurality of spiral grooves extending inwardly in the direction of rotation from the periphery of the bearing surface where there is a source of lubricant. Rotation of one of the bearing members relative to the other causes the lubricant to be drawn into the spiral grooves. The pressure of the lubricant is increased in the spiral grooves and the lubricant flows out between the bearing surfaces under sufficiently high pressure to support the rotating bearing member and thus the rotor of the machine to which it is fixed.

A further problem arises in gas-lubricated journal bearings, particularly those to be used with a vertical shaft. Usually, in a journal bearing carrying a load, the axis of the journal is not co-linear with the axis of the bearing sleeve. As the speed of the journal is increased

a speed of rotation, sometimes called threshold speed, is reached at which the axis of the journal begins to rotate about the axis of the sleeve. This phenomenon is usually termed "whip". As the speed of rotation is increased still further, the displacement of the journal axis from the sleeve axis may become so large and the whipping so violent that the journal surface makes contact with the confronting sleeve surface. This is particularly true when the bearing is only lightly loaded and as the load is decreased toward zero at any particular speed, the violence of the whipping increases. Also, the amplitude of the displacement of the journal axis from the bearing sleeve axis increases as the speed of rotation increases. This phenomenon then places a severe limitation on the speed at which gas-lubricated journal bearings may be run and on the load they may carry.

Accordingly, it is an object of this invention to provide a journal bearing having little tendency to whip even at very high speeds of rotation of the journal.

Another object of the invention is to provide an improved journal bearing of high stability particularly suited for use with a gaseous lubricant.

A further object of the invention is to provide an improved journal bearing useful in a vertical position where conventional liquid or solid lubricants cannot be used.

A still further object is to provide a journal bearing having a high load-carrying capacity at high speeds of rotation even when lubricated with a gas.

Other objects of this invention will be in part obvious and in part pointed out hereinafter.

Many advantages and modifications of the present invention will become apparent in the following description wherein reference is made to the accompanying drawings. In these, the invention is shown embodied in a substantially enclosed motor-driven centrifugal compressor unit employing a motor of the type wherein the rotor surrounds the motorstator. In the drawings:

FIG. 1 is a vertical axial section of such a motor-driven centrifugal compressor including two journal bearings fabricated in accordance with the present invention.

FIG. 2 is an elevation of the shaft shown in FIG. 1 and illustrates a journal surface in accordance with the invention.

FIG. 3 is a developed cross section of a portion of the shaft taken on the line 3—3 of FIG. 2.

FIG. 4 is a curve showing the relation between the load carried by an embodiment of the bearing and the minimum clearance between the journal surface and the confronting surface of the bearing sleeve at two different speeds.

FIG. 5 is a curve showing the relation for a particular embodiment of the invention between the ambient pressure of the gaseous lubricant and the minimum clearance which is maintained between the bearing surfaces at various loads placed on the rotor.

It will be observed that in the following discussion reference is made only to the use of gaseous lubricants. It will become evident, however, as the description proceeds, that liquid lubricants may also be used advantageously with the bearing provided by the present invention.

Referring to the drawings and particularly to FIG. 1, there is shown a substantially enclosed motor-blower unit comprising a sealed casing 10 and a rotor 11 adapted to rotate about a vertical axis. Rotor 11 com-

prises a hollow open-ended shaft **12** situated in a central cavity or well **13** and a pump impeller **14** situated in the upper part of the casing.

The vertical thrust of the rotor **11** is carried by a thrust bearing **15** of which the thrust collar **16** is integral with the shaft for rotation. Cooperating with the thrust collar **16** is the thrust plate **17** fixed to the casing **10.** The upper surface **18** of the thrust plate **17** may have spiral grooves as disclosed in the Dunning et al. application referred to, while the lower surface **19** of the thrust collar **16** may be a plane surface.

The gas to be pumped is drawn into the casing through a central flanged inlet **20** and flows through the vanes **21** of the impeller **14** into a discharge passage **24.** The impeller **14** and the discharge passage **24** cooperate to compress and pump the gas flowing therethrough.

The bottom of the well **13** has a re-entrant wall **22** which forms a central enclosure **23** within the open end of the hollow shaft **12.** The re-entrant wall comprises a thin cylindrical diaphragm **25** which is sealed at its lower end to the bottom of the well **13** and a disc **26** to which the upper end of the diaphragm is sealed. The enclosure **23** is closed at its lower end by a block **27** secured in substantially fluid-tight relation to the base of the casing, as by bolts **28.**

The rotor **11** is driven by means of a motor **29** comprising a motor-stator **30** mounted within the central enclosure **23** and a motor-rotor **31** mounted on the inside surface of the hollow rotor shaft **12.** (The terms motor-rotor and motor-stator are used to designate the electrically active rotating and stationary portions, respectively, of the motor.) The motor-stator **30** is supported on a centrally located hollow column **32** whose flanged lower end rests on the closure block **27** and is fixed thereto. The motor-stator **30** may have conventional field windings. The motor-rotor **31** comprises a plurality of rotor bars **33** fixed to the interior surface of the hollow shaft.

The diaphragm **25** is principally a protective measure, since it prevents the gas being handled by the pump from coming into contact with the motor-stator. However, in order to reduce electrical losses, the diaphragm **25** is preferably made of a metal having a high electrical resistance, such as Monel, or of a suitable non-conductor. The central hollow column **32** supporting the motor-stator **30** also provides support for the disc **26** and thus for the diaphragm **25.** In some cases, for example where the pumped fluid is non-corrosive, the diaphragm **25** may be omitted.

In order to cool the stator windings, a suitable coolant such as an oil may be flowed into the central enclosure **23** by means of the conduit **34** through the base block **27.** The coolant flows upward, surrounding the stator windings, and thence into openings **35** in the upper end of the central hollow column **32.** The coolant flows through the central channel **36** in the hollow column **32** and thence outward through conduit **37.**

The rotor receives its principal vertical support from the thrust bearing **15** and is guided laterally by the journal bearings **38** and **55.** The bearings **38** and **55** are situated adjacent each other and each comprises a stationary bearing member **39** having an inner plain bearing surface and a rotatable bearing member **40** having a confronting cylindrical surface formed in accordance with the present invention. These journal bearings support the radial thrust of the rotor **11.** It is seen here that the shaft **12** is formed with external bearing surfaces and the shaft itself acts as the journals of the two journal bearings.

Referring now particularly to FIG. **2,** there is shown in elevation the exterior surface of the shaft **12** having the two adjacent journal surfaces **46** and **47** formed in accordance with this invention, corresponding to the bearings **38** and **55** respectively. As shown, the shaft **12** is rotating in a counterclockwise direction when viewed from above the shaft. Each of the bearing surfaces **46** and **47** is substantially identical to the other so that the remarks made with respect to the bearing surface **46** will apply equally to the bearing surface **47.**

The bearing surface **46** is divided into an upper portion **56** and a lower portion **57.** The lower portion **57** is substantially a mirror image of the portion **56** with the mirror at the center of an annular peripheral band **53** encircling the bearing area **46** at its middle. In each of the upper portion **56** and the lower portion **57** of the bearing surface of the journal, a plurality of elongated depressed surfaces or recesses **42** and **42***a* respectively are arranged obliquely with respect to one another. The longitudinal edges of each depressed surface are parallel and the recesses **42** and **42***a* in portions **56** and **57** respectively diverge from their proximal ends at the central band **53** in the direction of rotation of the shaft. The distal ends of the depressed surfaces then lead the proximal ends with respect to the rotation of the shaft.

The depressions **42** may be formed by etching and are generally of the order of 0.0025 inches in depth when a gas is used as lubricant. Each depressed surface **42** is surrounded by the non-recessed portions **51** of the journal surface, the central band **53** being in the same surface as the non-recessed portions **51.** As shown in FIG. **3** wherein the proportions are exaggerated depressions **42** form a step-like rise **43** with the non-recessed portions **51** of the bearing surface and with the central band **53.** It will be seen then, since the lower portion **57** of the surface **46** is a mirror image of the upper portion **56,** that the non-recessed surfaces **51** form a herringbone pattern covering the entire area **46** of the journal, the central rib of the herringbone pattern being the central band **53.** It will also be seen that each of the depressed surfaces **42** has directly below it a corresponding depressed surface **42***a* in the lower portion **57** of the journal area. Each pair of depressions **42** and **42***a*, one in the upper portion **56,** the other its mirror image in the lower portion **57,** will be referred to hereinafter as a bearing element.

In the embodiment shown in FIG. **2,** the horizontal projection of each bearing element subtends a radial angle of approximately 60°, the projection being taken between the most remote corners of the recesses **42.** That is, the horizontal projection of each bearing element is approximately one-sixth the perimeter of the journal. Although this radial angle may have any value greater than zero and less than 360°, it is preferred that its value be less than 90° or one-fourth the perimeter of the journal.

The ratio of the area of the journal surface depressed to that which is not depressed is preferably between 0.25 and 2.0 although the value of this ratio depends largely on the desired characteristics of the bearing, particularly with respect to the load carried by the bearing and the running speed at which no appreciable whip is permissible.

The angle between the edges of the recesses and the central band **53** affects both the load-carrying capacity of the rotor and the tendency of the rotor to whip. As

this angle is made larger, that is, as the longitudinal edges of the depressions become more nearly perpendicular to the central band **53**, the tendency for the journal to whip is decreased. However, as this is done, the load-carrying capacity of the bearing is also decreased. Thus the two effects must be balanced against each other in designing a bearing of this type. It is necessary, however, that the edge of each depression trailing with respect to the direction of rotation be oblique to the central band **53** and to the axis of rotation. When fabricated in this way, the trailing step-like rise **43** is employed to cause the lubricant to flow longitudinally through the depression while at the same time the trailing rise **43** impedes the lateral flow of the lubricant out of the depression. The most desirable bearing is fabricated as shown in FIG. **2** with each bearing element consisting of a pair of elongated depressed surfaces obliquely arranged with respect to one another to diverge from one another in the direction of rotation. In this way rotation of the journal causes a flow of lubricant along the depressed surfaces from their distal ends towards their proximal ends.

It may be seen from the preceding description that the journal of each bearing has a plurality of bearing elements in spaced relation around the periphery of the journal, each element being formed by a pair of relatively shallowly depressed surfaces such as **42** and **42***a* with each depressed surface in an element being situated directly below and diverging from the other along the length of the journal surface. This in effect forms a herringbone pattern. It will be seen further that the journal surfaces **46** and **47** are substantially identical.

In order to reduce the starting torque in a compressor of this type, a gas under pressure may be admitted to the central well **13** by means of the conduit **50.** The pressure of this gas is sufficient just to lift the rotor and the gas is supplied only until the thrust bearing **15** is rotating at a sufficiently high speed to provide sufficient lubricant pressure to support the rotor. Gas supplied for this purpose may be either an inert gas or a portion of the gas being handled by the compressor.

As the operation of the compressor unit described above is started, the auxiliary gas under pressure is initially supplied to the well **13** through the conduit **50.** The pressure of this gas lifts the rotor to separate the thrust bearing surfaces **18** and **19.** As the rotor accelerates, gas is drawn in through the central inlet **20** into the pump impeller and this gas is pumped outward through the passage **24** by the impeller vanes. A portion of this gas surrounds the periphery of the thrust bearing **15** and is drawn in between the bearing surfaces by the rotation of the thrust collar **16** over the thrust plate **17.** This gas is pumped into the well **13** by the action of the thrust bearing and surrounds the rotor and the enclosure **23.** After the thrust bearing develops sufficient pressure to support the rotor, the auxiliary gas supply is shut off. As the pressure of the gas in the well is increased, a point is reached at which no further gas is pumped into the well so that all of the gas being pumped by the thrust bearing surfaces leaks out between the thrust plate and the thrust collar. There is then established a pressure drop across the thrust bearing with the maximum pressure being in the well, that is at the inner periphery of the annular thrust plate **17.**

The rotation of the shaft causes the gas in the well at the outer ends of the bearing to flow into and through the depressions **42** and **42***a*. The trailing edges of the pair of depressions in a bearing element, being obliquely arranged with respect to one another to diverge in the direction of rotation, cause a portion of the lubricant in the well to flow longitudinally along the depressions from their distal ends to their proximal ends. Thus each bearing element serves to distribute the lubricant along the length of the bearing area. The flow of the lubricant from the depressions is impeded by edges of the non-recessed area, that is, the step-like rises **43** trailing with respect to the flow of lubricant. This causes the gas flowing through the recesses to be compressed between the depressed surfaces **42** and the plain bearing surface **41** of the bearing sleeve. Thus the pressure of the gas in each depression **42** increases toward the trailing edge thereof, and the pressure of the gas is at its maximum at the trailing edges of the depressions. In this manner, the journal surface also acts as a pump to distribute the gaseous lubricant and to increase its pressure above the ambient pressure in the well **13** in a definite pattern around the periphery of the journal. It should be pointed out here that the depth of the depressions should never be large enough to permit the pressure of the lubricant to approach closely the value of the ambient pressure of the lubricant in the well. This would reduce the load-carrying capacity of the bearing markedly. In general, it is desirable that the depth of the depressed surfaces be of the order of magnitude of the machined radial clearance of the journal bearing.

The characteristics of a typical embodiment of the invention are shown in FIGS. **4** and **5.** In this embodiment two adjacent bearings were provided as described with each journal being 5.3 inches in diameter by 4.625 inches in length. The recessed areas were about 0.0024 inches in depth, the horizontal projection of each pair of recessed areas or bearing elements subtended a radial angle of about 60°. The journal bearing area was etched to form **24** bearing elements or pairs of recessed surfaces so that the recesses occupied about 20% more area than the non-recessed portions. The machined diametral clearance of the bearing was 0.0063 inches and at this clearance the bearing had a threshold whip speed at no load of about 7,000 r.p.m. However, when the maximum diametral clearance was reduced to about 0.005 inches, there was no whipping of the journal at 11,000 r.p.m.

In FIG. **4,** the clearance-load characteristics of the shaft just described are shown. The shaft was run with air lubrication at approximately atmospheric pressure, the curve A being made at 7,000 r.p.m. and the curve B at 6,300 r.p.m. The clearance indicated in the curve is the clearance at the point where the journal approaches the confronting surface of the bearing sleeve most closely, that is, at the point of minimum clearance. As the load on the rotor is increased the minimum clearance continually decreases with the higher load being carried at the higher speed of rotation at any particular minimum clearance. As indicated in the figure, even at a load on the bearing in excess of 50 pounds with a speed of rotation of 6,300 r.p.m., the clearance is around 0.001 inches. At 7,000 r.p.m. and a load of about 55 pounds the clearance is in excess of 0.0012 inches. As the load on the bearing is decreased at any speed below the no-load threshold speed, the minimum clearance approaches the machined clearance of the bearing.

In FIG. **5,** there are shown three curves demonstrating the effect of the ambient pressure of the lubricant, in this case air, on the minimum clearance as defined in

the preceding paragraph. Each of the curves was made with a different load on the same shaft, the curve A with a load of 13.32 pounds, B with 22.94 pounds and C with 32.38 pounds. As shown, the minimum clearance increases rapidly at each load up to a pressure of around 3 to 4 pounds per square inch absolute. Above this pressure the rate of increase of the clearance declines sharply so that the ambient pressure has virtually no effect on the minimum clearance once the ambient pressure exceeds about 4 pounds per square inch absolute. This indicates that the bearing described above should preferably be run with the ambient lubricant pressure in excess of about 4 pounds per square inch absolute. The effect of reducing the minimum clearance caused by increasing the load is also evident in FIG. 4.

Although the present invention has been described with reference to the use of a gas as lubricant it will be evident that the invention is also useful where liquid lubricants are employed. In the latter instance the threshold speed of rotation will be considerably increased so that the bearings may be used at even higher speeds with liquid than with gaseous lubricants without whipping. In addition, with liquid lubricants the load-carrying capacity of the bearing is markedly increased at any particular speed of rotation. It will be seen also that although the present invention has been described with reference to vertical bearings it may also be used in a bearing having a horizontal axis of rotation with either gaseous or liquid lubricants.

In the description of this invention the bearing elements, that is the pairs of recessed surfaces, have been provided on the rotating bearing member of journal. The bearing elements may also be formed in the stationary bearing member or bearing sleeve with the journal having a plain bearing surface. When this is done, however, the threshold whip speed of the bearing is reduced.

Other alternatives will be apparent to those skilled in the art.

Since many embodiments might be made of the present invention and since many changes might be made in the embodiment described, it is to be understood that the foregoing description is to be interpreted as illustrative only and not in a limiting sense.

We claim:

1. An improved journal bearing comprising in combination a non-rotatable cylindrical bearing member having a first bearing surface, a rotatable cylindrical bearing member having a confronting second bearing surface having a plurality of bearing elements, a source of lubricant adjacent said bearing elements for supplying lubricant thereto, each bearing element consisting of a pair of elongated relatively shallowly depressed surfaces lying in a cylindrical surface co-axial with the non-depressed surface and diverging from one another in the direction of rotation and obliquely arranged with respect to the axis of rotation of said rotatable member to cause a flow of lubricant longitudinally along said depressed surfaces from their distal ends toward their proximal ends as said bearing members are rotated relative to one another, each depressed surface subtending a radial angle of less than 360°, and means for rotating said rotatable bearing member to cause the lubricant to flow across and along said depressed surfaces, the flow of lubricant being impeded by the non-depressed portions of said second bearing surface to cause an increase in the lubricant pressure.

2. An improved journal bearing comprising in combination a hollow non-rotatable cylindrical bearing member having a plain inner bearing surface, a rotatable cylindrical bearing member with a confronting outer bearing surface having a plurality of bearing elements, a source of lubricant adjacent said bearing elements for supplying lubricant thereto, each bearing element consisting of a pair of elongated relatively shallowly depressed bearing surfaces lying in a cylindrical surface co-axial with the non-depressed surface and diverging from one another in the direction of rotation of said rotatable bearing member and obliquely arranged with respect to the axis of rotation of said rotatable member in a herringbone pattern to cause a flow of lubricant longitudinally along said depressed surfaces from their distal ends toward their proximal ends as said rotatable bearing member is rotated, each depressed surface subtending a radial angle of less than 360°, and means for rotating said rotatable bearing member to cause the lubricant to flow across and along said depressed surfaces, the flow of lubricant being impeded by the non-depressed portions of said outer bearing surface to cause an increase in the lubricant pressure.

3. An improved journal bearing comprising in combination a hollow non-rotatable cylindrical bearing member having a plain inner bearing surface, a rotatable cylindrical bearing member having a confronting outer bearing surface having a plurality of bearing elements, a source of lubricant adjacent said bearing elements for supplying lubricant thereto, each bearing element consisting of a pair of elongated relatively shallowly depressed surfaces lying in a cylindrical surface co-axial with the non-depressed surface and diverging from one another in the direction of rotation of said rotatable bearing member and arranged obliquely with respect to the axis of rotation of said rotatable member to cause a flow of lubricant longitudinally along said depressed surfaces from their distal ends toward their proximal ends as said rotatable bearing member is rotated, each said depressed surface having a projection in a plane perpendicular to the axis of rotation of said rotatable member subtending a radial angle of less than 360°, the ratio of the area of said outer bearing surface depressed to the area not depressed being in the range 0.25 to 2.0, and means for rotating said rotatable bearing member to cause the lubricant to flow across and along said depressed surfaces, the flow of lubricant being impeded by the non-depressed portions of said outer bearing surface to cause an increase in the lubricant pressure.

4. The improved journal bearing as in claim 3 wherein the projection of each bearing element in a plane perpendicular to the axis of rotation of said rotatable member subtends a radial angle of less than 90°.

5. An improved journal bearing comprising in combination a hollow non-rotatable cylindrical bearing member having a plain inner bearing surface, a rotatable cylindrical bearing member having a confronting outer bearing surface provided with a plurality of bearing elements, a source of lubricant adjacent said bearing elements for supplying lubricant thereto, each bearing element consisting of a pair of relatively shallowly depressed surfaces lying in a cylindrical surface co-axial with the non-depressed surface and diverging from one another in the direction of rotation of said rotatable member and arranged obliquely with respect to the axis of rotation of said rotatable member in a herringbone pattern to cause a flow of lubricant longitudinally along said depressed surfaces from their distal ends toward

their proximal ends when said rotatable bearing member is rotated, the total area of the depressed surfaces being between 33 and 60% of the area of said outer bearing surface, each said depressed surface subtending a radial angle of less than 360° with the depth of said depressed surfaces being of the same order of magnitude as the radial clearance between said rotatable and non-rotatable bearing members, and means for rotating said rotatable bearing member to cause the lubricant to flow across and along said depressed surfaces, the flow of lubricant being impeded by the non-depressed portions of said outer bearing surface to cause an increase in the lubricant pressure.

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